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(54) THREE PHASE ROTARY ENGINE

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TITLE

THREE PHASE ROTARY ENGINE

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ABSTRACT

The rotary engine employs two cylinders, a compressor cylinder and power cylinder, interconnected by a "loop" port. Rotors, with radial vanes, are disposed in the cylinders to define a plurality of working chambers. The compressor cylinder makes communication through an intake port with the ambient air. The compressor rotor on rotation draws in ambient air through the intake port and compresses the same in the compressor cylinder. The compressed air is fed through the loop port into the power cylinder when a combustion chamber of the power rotor is in registry with the loop port. During this time fuel is also injected into the combustion chamber. The combustion chamber indexes out of registry with the loop port to communicate with a firing cavity disposed in wall of the power cylinder. The mixuture of fuel and compressed ambient air is ignited by the firing cavity and the gases expand and vent through the exhaust port of the power cylinder, after having urged against the lead vane of the combustion chamber to rotate the power rotor. The gases flow through the engine in a smooth path following the shape of a figure eight in three cycles, intake, compression and combustion.

This invention relates to a rotary engine and particularly a rotary internal combustion engine.

Since the advent of the Wankel Rotary Engine there has been a rekindled interest in rotary internal combustion engines generally. The Wankel Engine has demonstrated distinct advantages over conventional two-and-four-stroke cycle reciprocating piston engines. These advantages include small physical size for a given horse power output; reduced number of components and moving parts; low mechanical friction, good mechanical balance, low octane requirements, reduced emissions of oxides of nitrogen, etc. These general advantages have not been without problems such as milling of epitrochoid chambers, sealing of the rotor, emission of unburned hydro-carbons on uncontrolled units, higher fuel consumption, spark plug fouling, metallurgical problems, and the difficulty of reboring worn cylinders. Furthermore, the duration of the power phase of a Wankel Engine is only approximately 60° of rotor rotation.

It is an object of the present invention to provide a rotary engine with the inherent advantages of piston rotary motion over piston reciprocating motion while at the same time extending the power phase beyond the 60° and preferably into the vicinity of 270° of each revolution of the rotary piston or of the power shaft. This not only results in improved thermal efficiency but also in superior emission control arising in part from the very long residence time of the burning gases.

It is a further object to eliminate construction of an epitrochoid chamber and therefore the eccentric motion of the Wankel rotor which results in complex gear-

ing and loss of space. This goal was reached by elongating the epitrochoid chamber of the Wankel engine to the point where two separate cylinders, divided by a narrow center wall are formed. Thus true rotary motion as opposed to eccentric motion is obtained, power is transmitted directly to the main shaft, the engine becomes even more compact and less complicated, and the air-fuel mixture follows the form of a figure-eight. Besides the beneficial effects of improved thermal efficiency and prolonged exhaust gas residence time, mentioned above, several other important advantages are derived from this extend flow pattern. The two main ones being:

(a) The conditions for each phase, especially in regard to temperature regulation, blow-by control and lubrication, can be easily maximized as they take place in different parts of the engine.

(b) It permits to increase thermal efficiency means of a very direct system of internal recuperation.

Another objective of the invention is to transmit gas pressures directly to main shaft of the engine without any intervening members such as crankshafts, connecting rods or eccentrics. This not only drastically simplifies overall design but based on effective displacement, as defined below, the engine of the present invention is roughly one-quarter the size of a Wankel engine or approximately one-eighth the size of a conventional piston engine.

In discussing further objects of the invent-

ion, it would be convenient to define certain terms; namely,

(a) "nominal displacement" shall be defined herein as the volume swept by one cylinder or working chamber during 1/2 revolution (180°) of the main shaft multiplied by the total number of power producing cylinders or working chambers,

(b) "effective displacement" shall be defined herein as the volume swept during all power strokes or power phases completed within two revolutions of the main shaft.

In the case of a piston engine the nominal displacement equals the effective displacement and both conform with the conventional meaning of the expression "engine displacement" as used in the automotive field.

It is an additional object of the present invention to provide a rotary engine wherein the effective displacement is twice the nominal displacement. This is made possible as the power phases of all working chambers formed by the vanes of each power rotor are completed without interruption within one, rather than two revolutions.

It is a further object to provide an engine which utilizes self-sustaining, continuous combustion, thus eliminating the need, during operation, of spark plugs, a distributor and the like. The main advantages derived from continuous combustion as opposed to intermittent combustion used by convential engines of the reciprocating and Wankel type are:

1. It permits the burning of extremely lean, stratified mixtures. Since engine speed under given load

conditions is controlled solcy by the amount of fuel injected, there is always an excess of oxygen present during combustion.

2. Thermal efficiency is increased due to improved combustion and the absence of temperature fluctuations. The latter is avoided as the other phases are placed conveniently into other parts of the engine.

3. The engine can be run on a variety of unleaded fuels.

The combined, above points result in very low emission, specifically as regards hydrocarbons, CO, oxides of nitrogen and lead salts. Typically all other types of gasoline engines produce significant amounts of at least one of above pollutants unless equipped with external emission control devices.

The invention achieves a gas flow pattern in the shape of a figure eight by employing two cylinder chambers having longitudinal axis essentially parallel to one another, and in the same plane a nearly tangentially disposed communication "loop" posrt or ports between said chambers, one rotor eccentrically disposed within each of the two cylinders, one rotor providing for the intake and compression phases, the other one for the two-stage power phase.

The invention therefore contemplates, a prime mover including:

(a) a housing defining first and second cavities with walls, a loop port communication with the cavities, an exhaust port communicating with the second cavity, and an intake port communicating with

the first cavity;

(b) an exantric compressor rotor mounted on a shaft disposed for rotation in said first cavity including sealing means bearing against the wall of the first cavity defining between the rotor, sealing means, and the walls a compression chamber of variable size according to the relative angular position of the rotor;

(c) an excentric power rotor mounted on a shaft disposed for rotation in the second cavity including sealing means bearing against the wall of the second cavity defining between the rotor, sealing means and wall, a power chamber of variable size according to the relative angular position of the rotor;

(d) a means for rotating the compressor rotor to draw air through the intake port and to compress the same within the varying sized chambers within the cavity;

(e) a means for injecting a combustionable fuel into the combustion chamber when said combustion chamber is in registration with the "loop" port, whereby further rotation of the power rotor causes said combustion chamber to pass out of registry with the "loop" port and injection device and move into registry with a firing region and for igniting the combustionable mix-

ture thereat whereupon the expanding gases created thereby urge against the surfaces of the vanes projecting out of the power rotor.

The invention will now be described by way of example with references to the accompanying drawings in which:

Figure 1 is a cross section through the prime mover.

Figure 2 is a perspective of the component parts, in assembly, of the prime mover of Figure 1.

Figure 3 is the gas flow pattern.

Figures 4, 5, 6 and 7 are sectional views along lines IV-IV, V-V, VI-VI, VII - V11 respectively.

Figures 8 and 9 are fragmentary section of and exhaust brake shown in the "off" and "on" position respectively.

Figure 10 is a fragmentary section of the spark plug configuration.

Figure 11 is an exploded perspective view of the vane detail of a rotor.

Figure 12, is the plan view of a rotor.

Figure 13, 14, 15 and 16 are respectively typical sections through the rotor respectively along lines X111-X111, X1V-X1V, XV-XV and XV1-XV1 of Figure 12.

Figure 17 is an exploded perspective of a heat exchanger.

Referring to Figures 1 and 2 the prime mover 20 includes a cylinder block 21 defining two cylinder chambers, a compressor cylinder 22 and a power cylinder 23; a plurality of nearly tangentially disposed "loop" ports 24

communicate between said cylinders.

The compressor cylinder 22 forms an enclosure for an eccentrically mounted rotor assembly consisting of a compressor rotor 26 having eight spring loaded radially extended vanes 27 which provide at their tips a sealing contact against the walls of the compressor cylinder 22 to define between said walls and the compressor rotor 26 eight oblong shaped segment chambers of varying size. Similarly the power cylinder 23 forms an enclosure for an eccentrically mounted rotor assembly consisting of a power rotor 25 also having eight spring loaded radially extended vanes 27 defining eight oblong shaped segment chambers of varying size.

The compressor rotor 26 and power rotor 25 are respectively mounted for rotation on a compressor rotor shaft 33 and a power shaft 34, having journal 35 supportable by suitable bearings 36 and 37 disposed in front and rear engine plates 38 and 39 as clearly seen in Figure 2. The journals 35 are equipped with journal grooves 40 serving as a receptacles for a journal seals 42 and expander rings 41 urging against said journal seals so that the latter are held firmly against the engine end plates 38 and 39.

Referring to Figures 11 and 12, each vane 27 is loosely mounted within a radial rotor slot 45. For each there are eight vanes arranged diametrically to one another. Each vane is equipped with two guide pins 46 over which a pair of coil springs 47 is mounted and shared with that of the diametrically opposed vane. The sharing of the coil springs offers more uniform pressure against the diametri-

cally opposed vanes since the distances between their tips or outer sealing surfaces varies only slightly at any angular position of a rotor within the cylinder as can be seen in Figure 1.

Each vane 27 is provided on its three outer edges with a vane groove 48 into which expander springs 49 are mounted and over which vane side seals 50 and a vane tip seal 51 sit. The vane seals 50 and 51 urge against the walls of the respective cylinders to provide eight oblong working chambers respectively between the vanes, the cylinder, and the body portion of the rotor. Sealing is enhanced between the rotor sectors 28 by the journal seals 42 inserted into the rotor journals 35 as can be seen in Figure 2. Preferably, a plurality of curved combustion chamber cavities 30 are disposed between the vanes in the perimeter of the power rotor 25 for purposes which will be described later.

The power shaft 34 is geared externally to the cylinder block or housing 21, through a power shaft gear 53 and a compressor shaft gear 52. Preferably the gear ratio is 2:1, and the compressor rotor 26 will therefore counter revolve at twice the speed of the power rotor 25, although any other gear ratio may be selected as may be convenient. The respective counter rotations of the rotors is assured by the direct coupling between the gears 52 and 53.

Referring to Figure 2 the engine end plates 38 and 39 are held to the main portion of the housing by transfer bolts 56 passing through suitably placed mounting holes 57 in the side of the engine assembly. The bolts may

be secured by nuts 58 at either the rear or front end plate.

The oblong working chambers vary in volume as the respective rotors revolve about their respective axis P_2 and C_2 which are arranged eccentrically to their respective cylindrical axis P_1 and C_1 of the power cylinder 23 and compression cylinder 22.

Referring to Figure 1, a fuel injection device 70, which may be of the conventional type, is mounted on the cylinder wall of the power cylinder 23, and in particular communicates with an expanding fuel injecting chamber 25, thereof defined by an itinerant sector 28 of the power rotor 25, and the respective vanes 27, since the power rotor rotates, and the circumferential arc of the adjacent cylinder wall. Also communicating with the power cylinder is a firing cavity 62 disposed within the cylinder wall an arcuat distance from the injection device 70. Cavity 62 may be a cupola in the cylinder wall communicating through a spark port 63 with a spark plug 60. The spark plug 60 is appropriately connected to a switched energizing source (not shown). The relative position of the firing cavity 62 and the fuel injection device 7 is such that the migrating fuel injection chamber 55 never communicates with the firing cavity 62 but solely with the fuel injection device 70 and loop ports 24.

Referring to Figure 3, the engine operates according to a "looped" three-phase cycle, and the gas flow throughout the engine follows that of an aero-dynamically smooth path in the shape of a figure eight, hence the term "looped three-phase cycle".

In operation the spark plug 60 is used solely for the purposes of initiating ignition. The initial starting or cranking cycle takes place as follows. Located in the rear end plate 30 are decompression valves 61A and 61B. The decompression valves are opened during the cranking cycle in order to reduce the load in the compression chambers of compressor cylinder 22. It should be noted that the torque requirements are alleviated, in part, by the 2:1 gear ratio between the power and compressor rotors 25 and 26. This permits the use of a free-wheeling starter drive (not shown) coupled directly to compressor shaft 33. When the engine has reached a sufficiently high cranking speed, the two decompression valves 61A and 61B are closed. At this point, the spark plug 60 introduces a spark (not shown) through the spark port 63, thereby initiating into the firing cavity 62 combustion in power cylinder 23. Details of the compression valves 61A and 61B, being common devices, are not fully disclosed as different forms may satisfactorily be used; so too with the spark ignition system and the starter drive 32.

Referring to Figures 1 and 3, the intake phase which occurs within the compressor cylinder 22, takes place in the upper half of the compressor cylinder. The compressor rotor 26 rotates clockwise causing the volume of the oblong working chambers in this part of the compressor, to increase. The volume of the chambers or chamber segments increases because of the eccentricity C_1-C_2 of the compressor rotor 26, hence the compressor vanes 27 extend during the intake phase. A low pressure region is created by this action, causing air to enter the compressor

cylinder 22 through an air intake or ram pipe 67 which communicates with the air cleaner or breather, generally indicated in Figure 2 as 68, from which ambient air is received. In order to facilitate free entry of the air throughout the intake phase, an arcuate intake channel or race 69 communicating with the intake port 67, is imbedded into the rear end plate 39 as clearly seen in Figures 1 and 2. The intake phase is completed foreach chamber segment when the trailing edge of the vane thereof passes over the end of the arcuate intake channel 69.

During the compression phase, occupying approximately the next 180° rotation of the compressor rotor 26, the volume of the working chambers decreases and the spring loaded vanes 27 are respectively pushed into their respective rotor slots 45. When a leading compressor vane reaches the loop or transfer ports 24, communicating directly between the compressor cylinder 22 and the power cylinder 23, the compressed air escapes into that combustion chamber of the power cylinder 23, which at this moment indexes with the transfer ports 24; that is into the expanding fuel injection chamber 55.

The transfer of compressed air from the compressor cylinder 22 to the power cylinder 23 occurs for the following reasons.

1. The compressor rotor 26 is fitted with very little clearance, and it has, as distinct from the power rotor 25, no rotor cavities 30;

2. The combustion chamber 55 which indexes with the transfer ports 24 is at a lower pressure than the compressor working chamber communicating with it;

3. The position and slant of the transfer ports 24 is so arranged that the momentum and centrifugal force of the rotating compressed air is fully utilized so as to eject it from the compressor cylinder 22;

4. The compressor rotor 26 which rotates at a higher speed than the power rotor has essentially properties analogous to a super charger;

5. Some of the compressed air escapes from the leading to the trailing combustion chamber segment during the very brief moment where both, transfer ports 24 and exhaust ports 79A are within the confines of two adjacent vanes forming the fuel injection chamber 55 being charged. For this reason the distance between the transfer ports 24 and the exhaust ports 79A is so chosen that the uncovering of the transfer ports 24 by the leading rotor vane of a given combustion chamber segment overlaps with that of the closing of the exhaust ports 75 by the trailing vane of the same chamber.

In summary, it may be stated that the overlap in port timing above described, is similar to the overlap of port timing on some two stroke cycle engines, and it serves essentially the same purpose. The overlap of the present invention serves specifically three functions:

1. it precharges the trailing combustion chambers;
2. it purges the trailing combustion chamber of any residual exhaust gases; and
3. it serves as an integral exhaust reactor by oxidizing unburned hydro-carbons,

this reduces the need of costly and power consuming emissi-

on control air pumps and air injectors employed on current engines.

The power or expansion phase of the engine takes place in the second cavity, power cylinder 23, in two connected stages and commences when a leading vane of the power rotor 25 transverses the firing cavity 62. When the engine is first started up, the firing is initiated by the spark plug 60, as earlier described, communicating with the firing cavity 62. The electrodes of the spark plug 60 are shielded against fouling by the small spark port 63.

Once the engine has fired the electrical ignition system (not shown) is turned off. At this stage combustion becomes self-sustaining and continuous as a controlled flame front passes through the firing cavity 62, and ignites the compressed volatile mixture in the trailing combustion chamber to the right of the firing cavity when referring to Figure 1. This occurs at that moment when the vane dividing the trailing chamber containing the compressed but not yet burning gases, and the leading chamber containing those already ignited, crosses the firing cavity 62. Thus the point of ignition is determined by the relative position of the firing cavity 62 within the power cylinder 23.

Once the mixture has been ignited, the high pressure of hot expanding gases pushes against the vanes 27 of the power rotor 25, and this causes the latter to turn in a counter clockwise direction.

The first stage of the power phase, which occupies almost 180° of revolution of the power rotor, is completed when the leading vane of a combustion chamber

segment uncovers the annular impeller or turbo ports 79. The turbo ports 79 which communicate directly with the exhaust ports 79A; are arcuate grooves or races in the perimeter of the cylinder wall of the power cylinder 23. Thus at this point the still expanding gases begin to by-pass the vane tips through these impeller ports in a controlled manner until they exit the power cylinder 23 via the exhaust ports 79A. The expanding hot gases continue therefore to exert a driving force against the vanes 27 of the power rotor 25 in the lower part of the power cylinder. This occurs despite the contraction of the combustion chamber segments taking place in this part of the power cylinder. The duration of the period in question, which constitutes the second stage of the power phase,is described by the length of the annular turbo ports 79. Escaping gases thus utilize the rotor vanes like turbine blades to provide a continuing driving force against them. This second stage of the power phase, which occupies approximately 90°, is completed when the leading vane of the given combustion chamber segment reaches the exhaust ports 79A.

Referring particularly to Figures 1, 5 and 17, the embodiment also provides direct recuperation by means of a heat exhanger 80. The very short exhaust ports 79A which form the terminal extension of the turbo ports 79 communicate directly with spill chamber 79B of the heat exchanger 80, and hence with the exhaust collector port 79C, and the exhaust pipe 79D attached externally to the cylinder block 21. The spill chamber 79B circumscribes the loop ports 24 and the hot gases,which pass through the exhaust ports 79A, thus preheat the compressed air passing through

the loop ports 24. This enhances thermal efficiency and emission control. The latter is derived from two factors,

1. improved vapourization and mixing of the fuel injected into the power cylinder;
2. increased residence time and oxidation of the exhuast gases.

The former is obtained by raising the temperature of the charge air which at this point is already fully compressed.

In order to provide improved engine braking under deceleration an exhaust brake 81 complete with valve member 82 is mounted in the exhaust collector port 79. By rotating the valve 82 from the position indicated in Figure 8, constituting normal operation, to that of Figure 9, full engine braking is obtained by simply rotating the rotary valve 82. This exhaust break system is similar to that employed with some diesel engines.

In order to ensure that the fuel air mixture is not ignited prematurely within the fuel injection chamber 55, the relative position of the firing cavity 62 is so selected that it may never communicate through a chamber segment with the fuel injection device.

The embodiments of the invention in which an exclusive property or privilige is claimed are defined as follows:

1. A prime mover including:

(a) a housing defining first and second cavities with walls, a loop port connecting first and second cavities for the purpose of transferring compressed air from the first cavity to the second cavity, an exhaust port communicating with the second cavity for the purpose of evacuating the burned gases from said second cavity, and an intake port communicating with the first cavity for the purpose of air intake;

(b) an eccentric compressor rotor mounted on a shaft disposed for rotation in said first cavity including sealing means bearing against the wall of the first cavity defining between the rotor, sealing means, and the walls a compression chamber of variable size according to the relative angular position of the rotor;

(c) an eccentric power rotor mounted on a shaft disposed for rotation in the second cavity including sealing means bearing against the wall of the second cavity defining between the rotor, sealing means and wall, a power chamber of variable size according to the relative angular position of the rotor;

(d) a means for rotating the compressor rotor to draw air through the intake port and to compress the same within the varying sized chambers within the cavity;

(e) a set of sliding vanes arranged radially in the eccentric power rotor, said vanes equipped with suitable seals bearing against the walls of the second cavity, thereby forming combustion chambers of variable volume,

B.

(f) a means of injecting a combustible fuel into the combustion chamber when said combustion chamber is in registration with the loop port, whereby further rotation of the power rotor causes said combustion chamber to pass out of registry with the loop port and injection device and move into registry with a firing region and for igniting the combustible mixture thereat whereupon the expanding gases created thereby urge against the surfaces of the vanes projecting out of the power rotor.

2. A prime mover including;

(a) a housing defining first and second cylinders, a loop port communicating with the cylinders, an exhaust port communicating with the second cylinder and an intake port communicating with the first cylinder;

(b) a compressor rotor mounted on a shaft eccentrically disposed for rotation in the first cylinder including radially arranged sliding vanes floating on the cylinder walls partitioning the cylinder into a plurality of compression chambers;

(c) a power rotor mounted on a power shaft eccentrically disposed for rotation in the second cylinder including vane surfaces floating on the cylinder walls partitioning the cylinder into a plural-

ity of combustion chambers;

(d) means gearing the compression and power shafts whereby the compressor rotor is rotated by the power shaft to cause a compression chamber communicating with the intake port to expand, to intake ambient gases through the intake port, and then to pass from communication with the intake port and to contract in size compressing the gases therein and to pass into registry with the loop port;

(e) means for injecting a combustionable fuel into a combustion chamber in registry with the loop port while compressed gases are received via the loop port from the first cylinder;

(f) means igniting the fuel mixture in the combustion chamber on its passing out of registry with the loop port whereby the result of the expanding ignited gases urge against the leading vane projections of the power rotor this causing the same to rotate, the gases continuing to expand and to be subsequently vented from the combustion chamber during communication of that chamber with the exhaust port.

3. The prime mover as claimed in Claim 2 wherein the power rotor rotates the power shaft.

4. The prime mover as claimed in Claim 2 wherein the rotors define a plurality of radial slots, a vane mount-

ed to float in each slot having sealing members about its parimeter with means urging the sealing members against the cylinders in order to define working chambers between the cylinder vanes and rotor.

5. The prime mover as claimed in claim 2 wherein the loop port is arranged in close proximity to the exhaust port.

6. The prime mover as claimed in claim 1 or 2 or 5 wherein the loop port is a plurality of ports.

7. The prime mover as claimed in claim 1 or 2 or 5 including a heat exchanger mounted in the path of the loop port.

8. The prime mover as claimed in claim 5 wherein the exhaust port communicates into a plurality of exhaust channels extending through the housing between the power and compression cylinders, and communicating with the power cylinder, and the loop port includes a plurality of loop ports extending across the exhaust channels whereby heat is transferred from exhaust channels to loop ports.

9. The prime mover as claimed in claim 2 or 4 including means in the exhaust channel for reducing the flow of the exhaust gases therethrough.

10. The prime mover as claimed in claim 1 including a firing cavity disposed, a determined distance from the loop port, in the wall of the second cavity, said determined distance being at least greater than the maximum length of an arc of the power chamber.

11. The prime mover as claimed in claim 10 wherein the determined distance is at least greater than the maximum length of an arc inscribed along the wall of the

power cylinder of a power chamber in registry with the loop port.

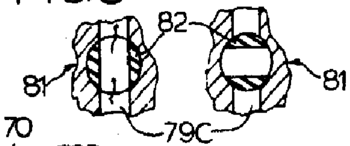
12. The prime mover as claimed in claim 2 wherein means (f) includes a firing cavity disposed, a determined distance from the loop port, in the wall of the power cylinder, said determined distance being greater in length then the maximum arcuate length of a power chamber.

13. The prime mover as claimed in Claim 2 wherein means (f) includes a firing cavity disposed, a determined distance from the loop port, in the wall of the power cylinder, said determined distance being at least greater then the maximum of the arc inscribed along the wall of the power cylinder of that power chamber in registry with the loop port.

14. The prime mover as claimed in Claim 12 or 13 including means communicating with the firing cavity to initiate combustion within the power cylinder.

FIG.10

FIG.8 FIG.9



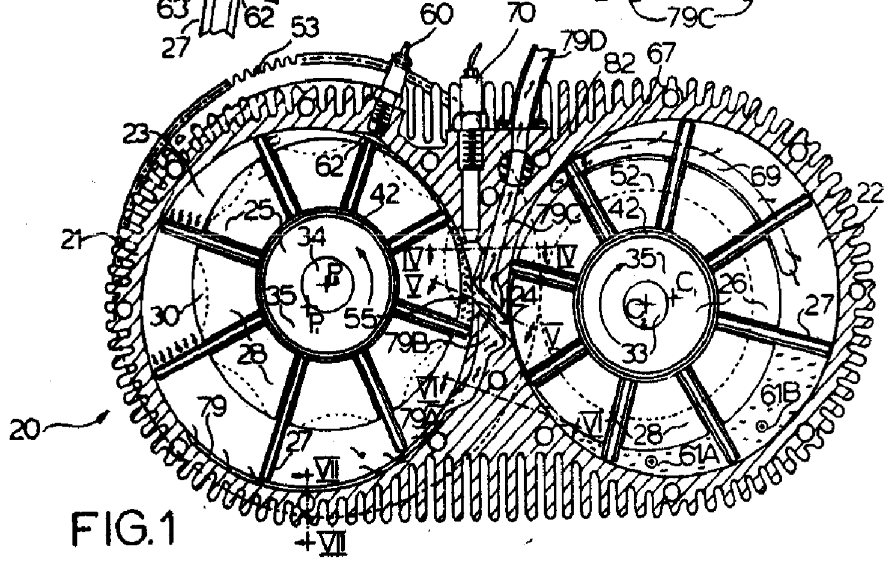


FIG.1

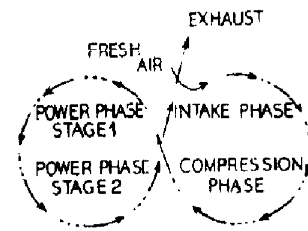
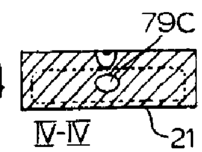
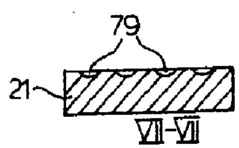
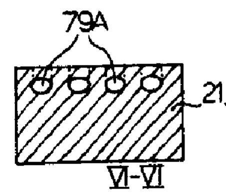


FIG. 3

FIG.4







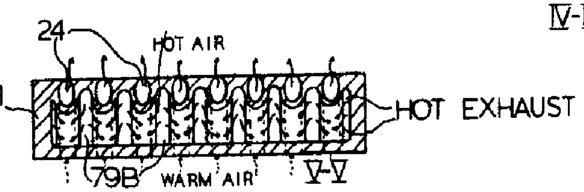
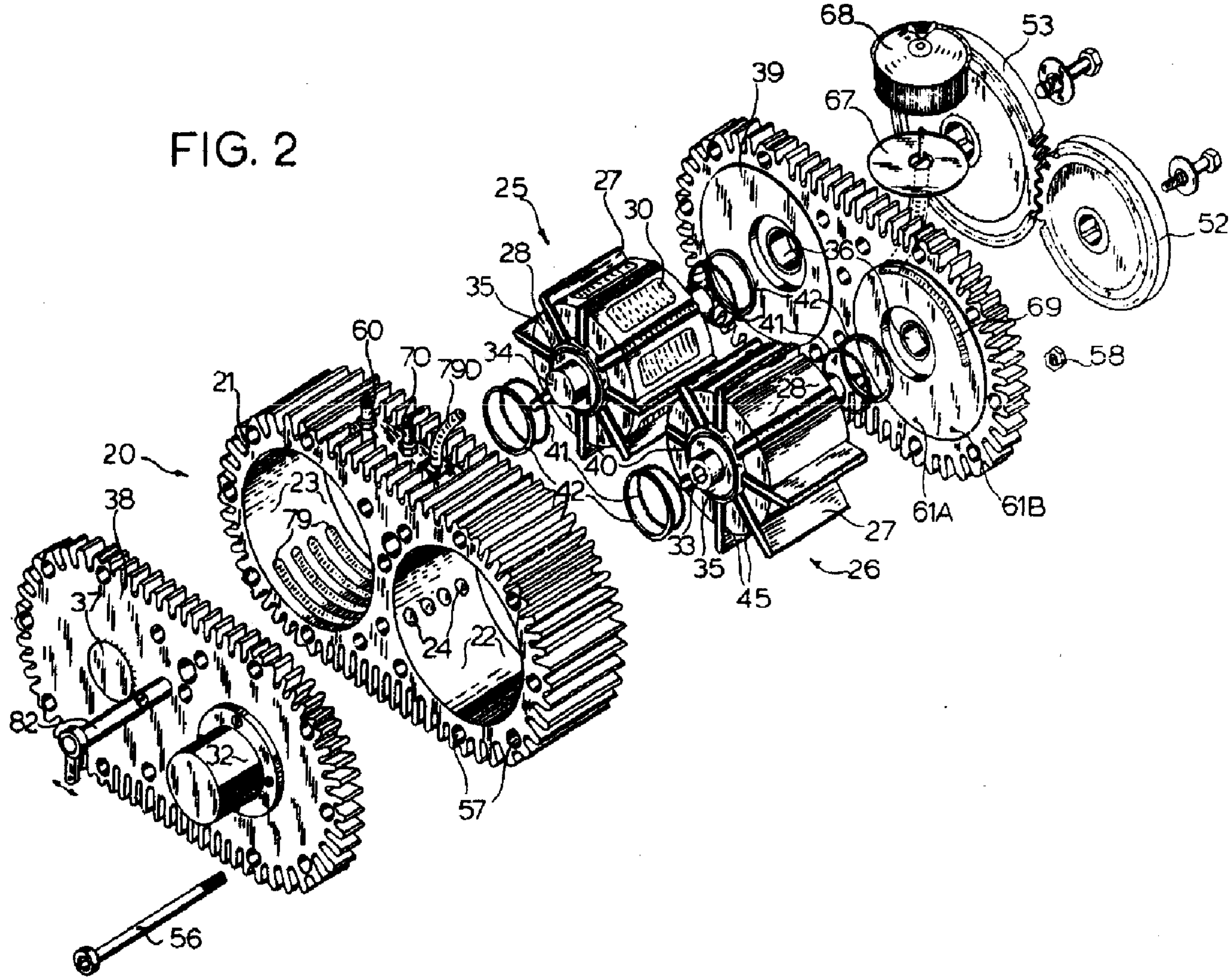


FIG.7 FIG.6 FIG.5

FIG. 2



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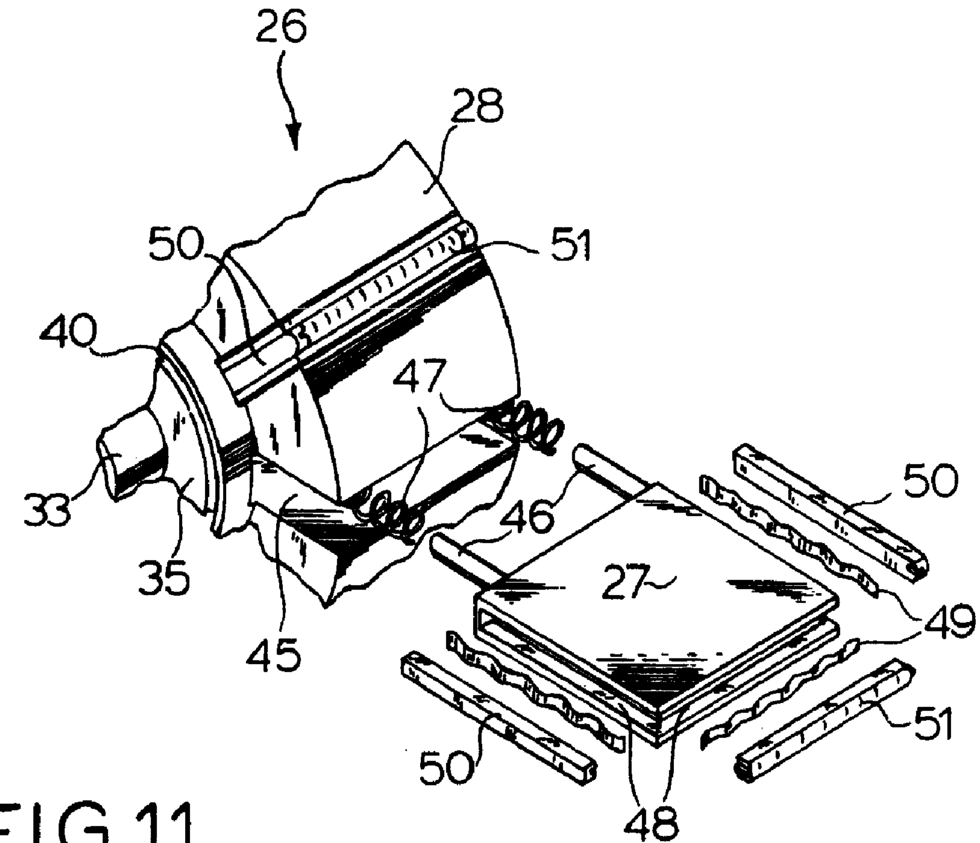


FIG.11

FIG.13 XIII-XIII

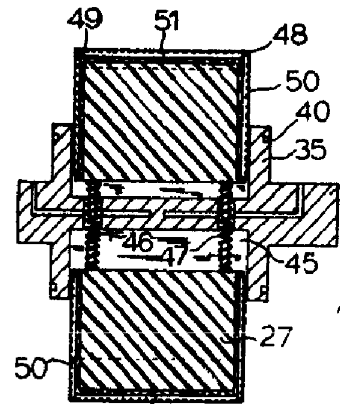
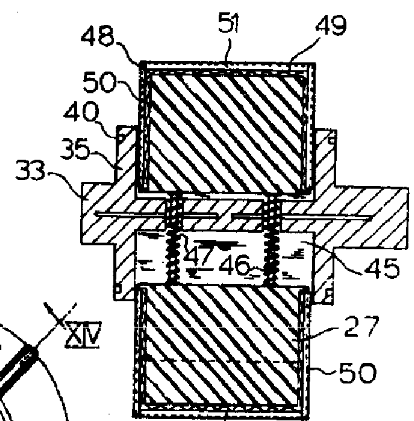


FIG.15 XV-XV



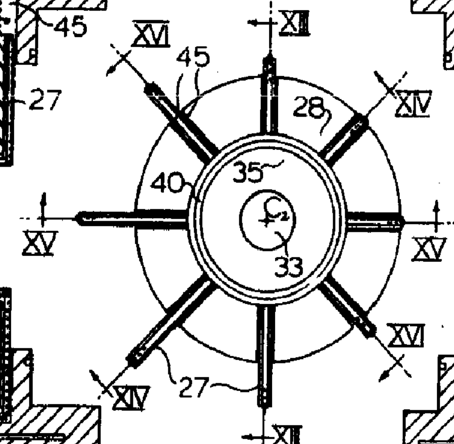


FIG.12

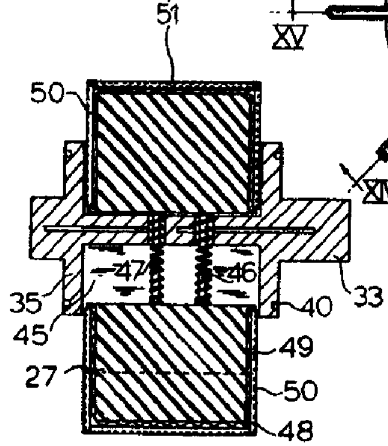


FIG.14 XIV-XIV

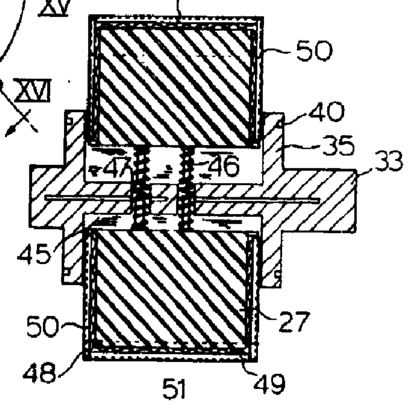
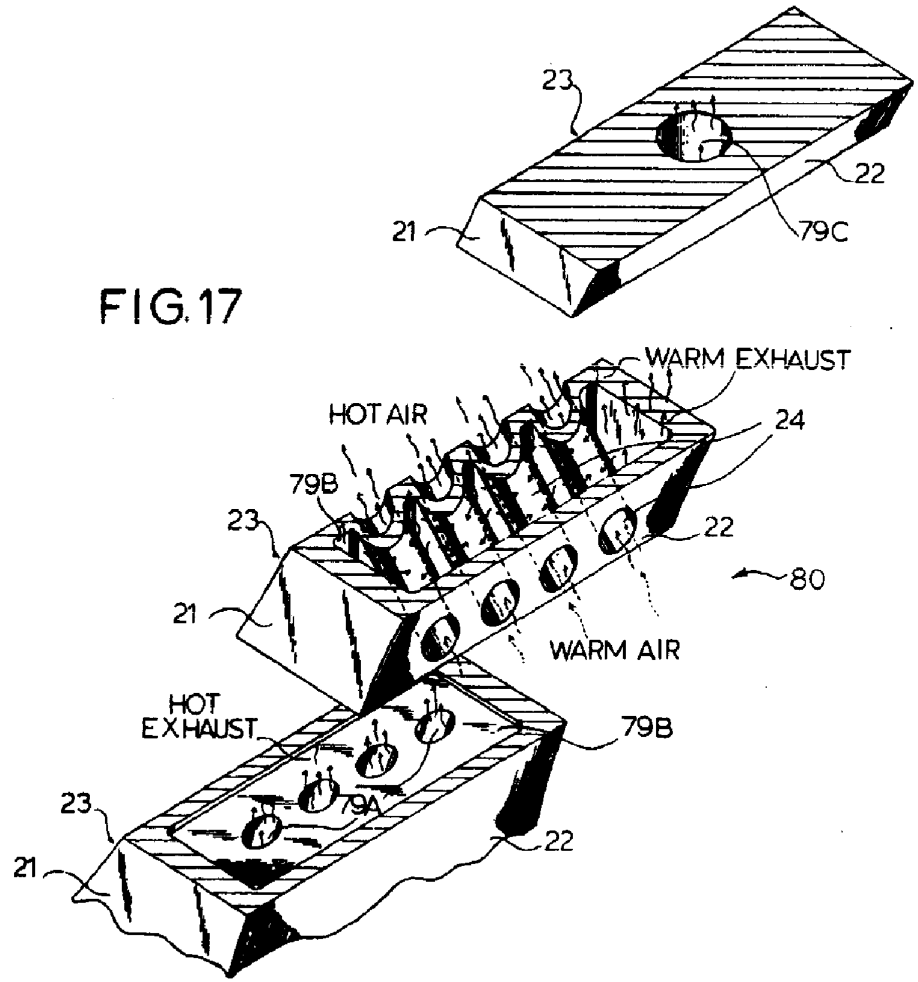


FIG.16 XVI-XVI

FIG.17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